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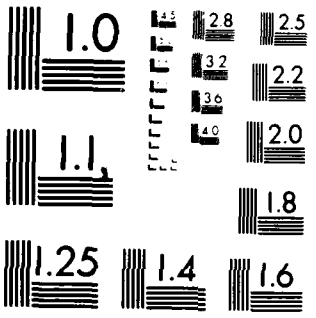

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YIELD SURFACE FOR BARS INCLUDING WARPING RESTRAINT

Weidlinger Associates
333 Seventh Avenue
New York, New York 10001

1 February 1982

Technical Report

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A procedure is described for deriving yield surface equations for thin walled open cross sections for cases in which Saint Venant torsion is small compared to warping torsion. The yield surface equations are expressed in terms of an axial force, bending moments in two planes, and the warping moment or bimoment. Equations are derived for the specific case of a z-section. These equations are shown to satisfy the uniqueness theorem of plastic analysis thereby —

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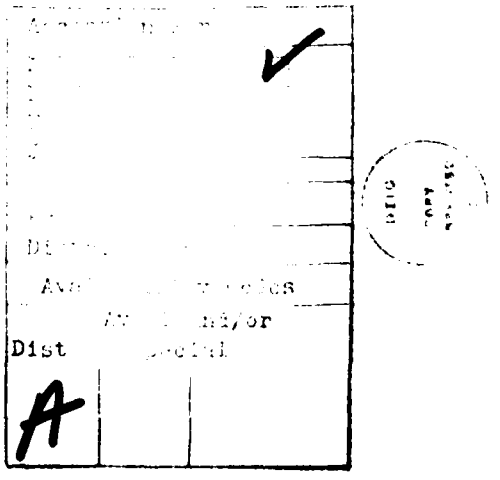
20. ABSTRACT (Continued)

furnishing the correct yield surface within the limits of validity of the assumptions made, The applicability and accuracy of the proposed equations are indicated.

PREFACE

The authors would like to thank Professor Atle Gjelsvik for his gracious assistance and numerous suggestions during the course of this work.



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INTRODUCTION

In recent years, considerable progress has been made in the development of computer programs to perform dynamic, large deflection, elasto-plastic analysis of submarine structures (1, 2, 3). A typical problem of interest is the dynamic elasto-plastic analysis of submerged, stiffened cylinders. Stiffeners on such cylinders are usually thin walled open cross sections such as T, Z, L or flat bar sections.

It is well known that if a section has an enforced center of rotation, (i.e. it is not free to twist about its shear center, as a stiffener welded to a shell is not), there is an increase in the warping resistance of the section to torsion (13, 22). For certain cases, it has been shown that the stresses resulting from resistance to warping can have considerable effect on member response and stability (12, 13, 23). In addition, experiments (7, 8, 19) have clearly indicated that importance of stiffener tripping (lateral-torsional instability) as a critical failure mode for ship structures. Hence, it becomes desirable to consider restrained warping stresses in the development of an elasto-plastic stiffener for inclusion in the analysis of stiffened cylinders.

The inclusion of warping stresses in the elastic response of thin walled open cross sections has been studied by many authors in great detail (13, 14, 22). The plastic response of the section under combined stresses is defined by a yield surface. Following procedures suggested by Hodge (16), Morris and Fenves (19) have determined an approximate lower bound to the exact yield surface for a variety of cross sectional types. This investigation included Saint Venant torsion but neglected warping restraint. Boulton (5), Dinno (9, 10) and Gjelsvik (13) have studied the effect of warping restraint on plastic collapse of thin walled open cross sections, but only for simplified cases in which one or more of the possible stress resultants are zero.

This report presents the formulation of an elasto-plastic thin walled open cross section when the effect of warping torsion is more important than Saint Venant torsion. Equations are developed for the yield surface of a Z-section in terms of four stress resultants, namely the axial force, bending moments in two planes, and the warping moment. These are shown to satisfy both upper and lower bound theorems of plasticity and therefore correspond to exact yield surfaces within the assumptions of thin walled sections.

The development of a yield surface including warping restraint is a necessary first step in the analysis of inelastic stiffener tripping. The equations are currently being incorporated into the computer program EPSA (2), developed at Weidlinger Associates under DNA/ONR sponsorship. This implementation is aimed at extending the modeling capability of EPSA to include stiffener tripping and improving structural response predictions in critical regions of large strain and deformation. This capability will be required in the analysis of a series of plastic range tests on both torpedo (7) and submarine scaled model (8) structures.

KINEMATICS OF THIN WALLED OPEN CROSS SECTIONS

The assumptions made in the formulation of the theory of thin walled beams are that cross sections deform as rigid bodies, and that the shear strain at the middle surface of the section is zero (13, 22).

Referring to Fig. 1, the axial displacement of a point on the middle surface of the bar, w(s, z), may be written as (13),

$$w(s, z) = W(z) - U'(z)\, x(s) - V'(z)\, y(s) - \phi'(z)\, \omega(s) \qquad (1)$$

where $\bar{n}$-s-z are coordinates with $\bar{n}$ normal to, and s following the contour of the middle surface; U(z), V(z), W(z) represent the translation of the center of rotation P in the x-y-z coordinate system; $\phi(z)$ is the rotation of the section about the pole P; $\omega(s)$ is the warping function defined as

$$\omega(s) = \int_0^s r(s)\, ds \qquad (2)$$

where r(s) is the perpendicular distance form the point P to the point in question on the middle surface. Thickness or secondary warping is neglected in Equation 2.

The first three terms on the right hand side of Equation 1 satisfy Navier's assumption while the last term represents warping.

Stress and Strain Components

The only non-zero stress components are the longitudinal stress, σ_{zz}, and the shearing stress, τ_{zs}, shown in Fig. 2. All stress and strain components normal to the wall are neglected. In addition, all in-plane strains are assumed small. From Eq. 1, the longitudinal strain

is

$$\varepsilon_{zz}(s,z) = W'(z) - U''(z)\, x(s) - V''(z)\, y(s) - \phi''(z)\, \omega(s) \quad (3)$$

If it is desired to extend this formulation to account for geometric nonlinearities, the appropriate nonlinear strain-displacement relations would have to be introduced. This has been done in (6).

Stress Resultants

The strain energy per unit length of the bar, U_s, can be expressed as (13),

$$2U_s = NW' + M_y U'' - M_x V'' + M_\omega \phi'' + T_s \phi' \quad (4)$$

In Eq. 4, the stress resultants N, M_y, and M_x represent the axial force and bending moments about the y-axis and x-axis, respectively, Saint Venant torque is denoted by T_s, and M_ω is the warping moment, or bimoment, and is defined as

$$M_\omega = - \int_A \sigma_{zz} \omega \, dA \quad (5)$$

These stress resultants are shown in Fig. 3.

DECOUPLING OF THE MIXED TORSION PROBLEM

The last two terms of the right hand side of Eq. 4 are the contribution of the warping moment and Saint Venant torsion to the strain energy of the bar. In certain applications, effects of one of these may predominate. In the elastic range, the relative importance of Saint Venant torsion to the warping torsion in straight beams is given by the magnitude of a dimensionless bar parameter, μ_L, defined in (13) as

$$\mu_L = L/d \tag{6}$$

in which L is the span length of the bar and d is a characteristic length defined by

$$d^2 = EI_{\omega\omega}/GJ \tag{7}$$

where E and G denote Young's modulus and the shear modulus and $I_{\omega\omega}$ and J are the warping constant and torsion constant.

It has been shown, (13 , 22), that when μ_L is small (approximately 1) warping torsion is dominant and the Saint Venant torsion term in Eq. 4 may be neglected. If μ_L is large (approximately 20), Saint Venant torsion controls and the term containing the warping moment in Eq. 4 may be dropped.

In the plastic range, the relative importance of these two effects may be estimated by using Eq. 6 with d assumed equal to its elastic value and L taken as an effective length corresponding to plastic deformation patterns of the bar.

Case Study

The preceding discussion on approximations to the mixed torsion problem is applied to a typical problem of interest. Consider the

ring-stiffened cylinder in Fig. 4. A cross section of a typical stiffener is also shown. The value for the warping constant, $I_{\omega\omega}$, for the Z-stiffener with an enforced center of rotation about the line of attachment between the stiffener and the shell is .0176 in^6 (4.73 x 10^6 mm^6). The value of the torsional constant, J, is .00253 in^4 (1053 mm^4), (14, 15). Using a value for E/G = 2.6 for the steel structure, d^2 given by Eq. 7 is 18 in^2 (11610 mm^2).

It has been shown experimentally and analytically in (4, 18) that for cylinders with relatively stiff ring-stiffeners, plastic deformational patterns are characterized by a large number of circumferential waves (typically greater than five). A conservative estimate of an upper bound to the effective length of the ring-stiffener may therefore be taken to be that corresponding to the n = 5 deformational mode. The resulting value of μ_L should also be an upper bound. The effective length is then one-tenth of the ring circumference. Using a value of 8.9 in (226 mm) for the length of the stiffener, a value of μ_L = 2 is obtained. With this value of μ_L as an upper bound, Saint Venant torsion may be neglected in Eq. 4, and only the four stress resultants contributing to the axial stress, σ_{zz}, need be considered.

PLASTIC BEHAVIOR

In the formulation of a plastic theory of thin walled bars, the assumption is made that the kinematics describing the plastic behavior of the bar are the same as in the elastic range, (13).

The plastic response of a bar is defined by a yield surface. The assumptions made in the derivation of this surface, in addition to the kinematic ones are the following:

1) the material is elastic-perfectly plastic,
2) the von Mises yield condition applies,
3) shear stresses due to warping are ignored as in plastic beam theory, (20),
4) the yield surface is a closed, convex surface and the incremental plastic deformation is directed outward, normal to the surface. With these assumptions, Drucker's hypotheses,(11), are satisfied insuring both uniqueness and stability of plastic response.

The validity of any proposed yield surface may be assessed using the upper and lower bound theorems of plastic analysis, (20). The lower bound theorem states that if a yield surface is derived from a safe and statically admissible stress state it will be enclosed by or coincide with the correct yield surface. The upper bound theorem states that if the states of stress on a yield surface correspond to assumed mechanisms, that yield surface encloses or coincides with the correct yield surface. These theorems may be combined to form a uniqueness theorem which states that a yield surface, derived from a

safe and statically admissible stress state, and corresponding to a kinematically admissible mechanism, is the one correct yield surface.

The yield condition, formulated in terms of normalized stress resultants, involves only those stress resultants which are contained in the expression for the strain energy, (16). Referring to the previous discussion on approximations in the mixed torsion problem, it is assumed that the effects of Saint Venant torsion are negligible. The strain energy of the bar, U_s, of Eq. 4, reduces to

$$2U_s = NW' + M_y U'' - M_x V'' + M_\omega \phi'' \tag{8}$$

Hence, the yield condition may be expressed in terms of the dimensionless parameters

$$n = N/N_p \tag{9a}$$

$$m_x = M_x/M_{xp} \tag{9b}$$

$$m_y = M_y/M_{yp} \tag{9c}$$

$$m_\omega = M_\omega/M_{\omega p} \tag{9d}$$

in which N, M_x, M_y, and M_ω are the previously defined stress resultants shown in Fig. 3 and N_p, M_{xp}, M_{yp}, $M_{\omega p}$ are their fully plastic values. Since shear stresses have been neglected, the von Mises yield condition reduces to

$$\sigma_{zz}^2 = \sigma_y^2 \tag{10}$$

in which σ_y is the yield stress of the material under uniaxial stress.

DETERMINATION OF A LOWER BOUND YIELD SURFACE

It is evident from Eq. 1 that, if warping is considered, Navier's hypothesis is not valid for the cross section as a whole. It is assumed, however, that Navier's hypothesis does hold for each element of the cross section. This implies that there is one neutral axis contained in each element of the cross section.

The equations for the yield surface may be simplified by the following approximations for thin walled sections (19);

1) the neutral axis passes perpendicularly through any element of the section, as shown in Fig. 5, or,
2) if the neutral axis lies wholly within an element, it is parallel to the long dimension of the element, Fig. 5.

To illustrate the derivation of yield surface equations for a thin walled open cross section including warping stresses, consider the Z-section in Fig. 6. The dimensions of the section which has an enforced center of rotation at P are:

b = width of the flanges;

h = depth of section, center to center of the flanges;

A_f = area of one flange; A_w = area of web;

t = thickness of web and flanges; $c = A_w/A_f = h/b$;

$\bar{h}$ = distance from P to center of top flange where typically $h/\bar{h} \simeq 1$.

The fully plastic stress resultants are found to be:

$$N_p = \sigma_y A_f (2 + c) \tag{11a}$$

$$M_{xp} = \frac{1}{4} \sigma_y A_f h (4 + c) \tag{11b}$$

$$M_{yp} = \sigma_y A_f b \tag{11c}$$

$$M_{\omega p} = \frac{1}{2} \sigma_y A_f b\bar{h} \tag{11d}$$

In general, a yield surface has several elements each related to generic neutral axis positions. The approximations regarding neutral axis location within an element allow a reduction in the number of surface elements comprising the yield surface of a Z-section to four. They are shown in Fig. 7. However, since the section is not symmetric, four complementary cases are necessary for complete determination of the yield surface. In these complementary cases, indicated in Fig. 8, the generic neutral axis locations of Fig. 7 are unchanged but the sign of the yield stress is reversed.

The stress resultants corresponding to the stress distribution of Fig. 9 are:

$$N = \sigma_y A_f(2 + c - 2(\beta + \gamma c + \alpha)) \tag{12a}$$

$$M_x = \sigma_y A_f h(\gamma c(\gamma - 1) - \alpha + \beta) \tag{12b}$$

$$M_y = \sigma_y A_f b(\beta^2 - 2\beta + \alpha^2) \tag{12c}$$

$$M_\omega = \frac{1}{2} \sigma_y A_f b\bar{h}(1 - 2\alpha^2) \tag{12d}$$

in which α, β, and γ are dimensionless parameters that define the neutral axis location in each element of the cross section.

The corresponding normalized stress resultants, found by substituting Eqs. 12 and 11 into Eqs. 9, are:

$$n = 1 - \frac{2(\beta + \gamma c + \alpha)}{2 + c} \tag{13a}$$

$$m_x = \frac{4}{4+c} (\gamma c(\gamma - 1) - \alpha + \beta) \tag{13b}$$

$$m_y = \beta^2 - 2\beta + \alpha^2 \tag{13c}$$

$$m_\omega = 1 - 2\alpha^2 \tag{13d}$$

Eliminating the parameters α, β, and γ from Eqs. 13 , the equation of the element of the yield surface for this case is obtained as

$$\left[\frac{2+c}{2}(1-n) - \left(1 - \sqrt{\tfrac{1}{2}(1+m_\omega) + m_y} + \sqrt{\tfrac{1}{2}(1-m_\omega)}\right)\right]$$

$$\times \left[\frac{2+c}{2c}(1-n) - \frac{1}{c}\left(1 - \sqrt{\tfrac{1}{2}(1-m_\omega) + m_y} + \sqrt{\tfrac{1}{2}(1-m_\omega)}\right) - 1\right] \quad (14)$$

$$+ 1 - \sqrt{\tfrac{1}{2}(1+m_\omega) + m_y} - \sqrt{\tfrac{1}{2}(1-m_\omega)} - \left(\frac{4+c}{4}\right) m_x = 0$$

Consider the stress distribution shown in Fig. 10. In this case, a neutral axis lies within the bottom flange. The values of n and m_x are insensitive to the angle θ since it is necessarily small. The normalized warping moment, m_ω, is independent of θ, and the bending moment about the y-axis, m_y, is an unknown function of θ and hence indeterminate. The normalized stress resultants are:

$$n = \frac{c + 2(-1 + \beta + \alpha)}{2 + c} \quad (15a)$$

$$m_x = \frac{4(\beta - \alpha)}{4 + c} \quad (15b)$$

$$m_y = \text{indeterminate} \quad (15c)$$

$$m_\omega = 1 - 2\alpha^2 \quad (15d)$$

The equation for this element of the yield surface, obtained from Eqs. 15 , is independent of m_y and may be written

$$(2 + c)\, n + \frac{4 + c}{2} m_x + \sqrt{\beta(1 - m_\omega)} + c - 2 = 0 \quad (16)$$

Similarly, equations for the other elements were found. They are shown in tabular form in Figs. 7 and 8. The yield surface constructed from the various surface elements is illustrated in Fig. 11 for the Z-stiffener of Fig. 4. A two dimensional section in the $n - m_x$ plane is shown for $m_y = m_\omega = 0$. The range of applicability of each surface

element is indicated on the figure. Isometric plots of the yield surface are shown in Figs. 12 and 13 for $m_y = 0$ and $m_x = 0$, respectively.

DETERMINATION OF UPPER BOUND - YIELD MECHANISMS - UNIQUENESS

The state of stress in the bar may be expressed in terms of the stress resultants, which may be considered components of a matrix denoted by {Q}:

$$\{Q\} = \{ N, M_y, M_x, M_\omega \} \tag{17}$$

The corresponding matrix of bar strains, {q}, is

$$\{q\} = \{ W', U'', -V'', \phi'' \} \tag{18}$$

The yield surface is a function of the stress resultants such that, at yield,

$$F (N, M_y, M_x, M_\omega) = 0 \tag{19}$$

It has been assumed that the incremental plastic strain is directed outward, normal to the yield surface. This associated flow rule may be stated as

$$\delta q_i = \lambda \frac{\partial F}{\partial Q_i} \tag{20}$$

in which λ is an arbitrary positive proportionality factor and δq_i denotes a component of incremental plastic strain.

From Eq. 20 it follows that the strain increments corresponding to stress points on the yield surface satisfy

$$\frac{\delta q_i}{\delta q_j} = - \frac{\partial Q_j}{\partial Q_i} \tag{21}$$

in which the derivatives on the right hand side are obtained from Eq. 19 .

To illustrate the determination of such strain increment rates, consider, e.g., the lower bound stress distribution of Fig. 10. The yield surface equation for this stress state is given by Eq. 16.

The ratios of the components of the plastic strain increments follow from Eqs. 11, 16, and 21. They are

$$-\frac{\delta V''}{\delta W'} = -\frac{\partial N}{\partial M_x} = \frac{2}{cb} \tag{22a}$$

$$\frac{\delta \phi''}{\delta W'} = -\frac{\partial N}{\partial M_\omega} = -\frac{2}{h\alpha b} \tag{22b}$$

$$\frac{\delta U''}{\delta W'} = -\frac{\partial N}{\partial M_y} = 0 \tag{22c}$$

Yield mechanisms may be determined solely from kinematic considerations. Following Eq. 3, the incremental plastic strain may be written as

$$\delta\varepsilon_{zz} = -\delta W'(-1 + \frac{\delta U''}{\delta W'} x + \frac{\delta V''}{\delta W'} y + \frac{\delta \phi''}{\delta W'} \omega) \tag{23}$$

At a neutral axis location in an element of the cross section, the axial strain, ε_{zz}, and hence, $\delta\varepsilon_{zz}$, are zero. The neutral axis locations for each surface element are known and indicated in Figs. 7 and 8. Therefore, the components of the yield mechanism may be determined.

Again, consider the stress distribution of Fig. 10. The neutral axis locations and hence the positions at which Eq. 23 equals zero are shown in Table 1. Substitution of the conditions specified in Table 1 into Eq. 23 gives the components of the yield mechanism as follows:

$$-\frac{\delta V''}{\delta W'} = \frac{2}{cb} \tag{24a}$$

$$\frac{\delta \phi''}{\delta W'} = -\frac{2}{h\alpha b} \tag{24b}$$

$$\frac{\delta U''}{\delta W'} = 0 \tag{24c}$$

Comparison of Eqs. 22 and 24 shows that the stress state on this element of the lower bound yield surface corresponds to a yield mechanism. Thus, the uniqueness theorem of plastic analysis is satisfied and the yield surface element, given by Eq. 16, is the correct one.

In a similar manner, the uniqueness theorem can be shown to be satisfied for the remaining surface elements of Figs. 7 and 8. Hence, the complete yield surface, defined by the individual surface elements, is the exact yield surface. The yield mechanisms, derived from kinematics, for each surface element are listed in Table 2.

ACCURACY AND IMPLEMENTATION OF THE PROPOSED FORMULATION

The accuracy of the proposed formulation rests on two approximations. The first involves the elimination of Saint Venant torsion from the yield condition. The second involves assumptions on neutral axis location within the elements of the cross section.

Referring to the earlier discussion on approximations to the mixed torsion problem, it is evident that as the importance of Saint Venant torsion increases, the proposed formulation will be increasingly in error.

The assumption that a neutral axis passes perpendicularly through an element of the cross section leads to artificial ridges in the yield surface. These ridges lie outside the correct yield surface. This approximation was discussed in detail in (19), and has been shown to be negligible in elasto-plastic analysis of thin walled sections.

The technique used in the development of the yield surface for a Z-section including warping stresses can be applied to any thin-walled open cross section. Other common types used as ring-stiffeners for cylindrical shells are T and L sections.

The proposed yield surface equations are suitable for incorporation into the existing nonlinear, elasto-plastic finite element program, EPSA (2). The present stiffener formulation in the code accounts for plastic response in terms of only two stress resultants, namely the axial force and bending moment about the strong axis of the stiffener. The proposed formulation will extend the structural modeling capability for the class of problems such as that shown

in Fig. 4, and lead to an increase in the accuracy of structural response predictions, especially in regions of large plastic strain and deformation.

CONCLUSION

A procedure has been described for deriving yield surface equations for thin-walled open cross sections with an enforced center of rotation for cases in which Saint Venant torsion is small compared to warping torsion. Equations have been derived for the specific case of a Z-section. These equations have been shown to satisfy the uniqueness theorem of plastic analysis thereby furnishing the correct yield surface within the limits of validity of the assumptions made. The applicability and accuracy of the proposed equations are indicated.

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NOTATION

The following symbols are used in this paper:

A_f = cross sectional area of flange;
b = width of flange;
c = ratio of area of web to area of flange;
d = characteristic length;
E = Young's modulus;
F = function defining yield surface
G = shear modulus
h = depth of section center to center of flanges;
$\bar{h}$ = distance from pole to center of top flange;
$I_{\omega\omega}$ = warping constant;
J = torsion constant;
L = length of bar;
M_x = bending moment about x-axis;
m_x = normalized bending moment about x-axis;
M_{xp} = fully plastic bending moment about x-axis;
M_y = bending moment about y-axis;
m_y = normalized bending moment about y-axis;
M_{yp} = fully plastic bending moment about y-axis;
M_ω = warping moment;
m_ω = normalized warping moment;
$M_{\omega p}$ = fully plastic warping moment;
N = axial force;
n = normalized axial force;
N_p = fully plastic axial force;
$\bar{n}$-s-z = coordinate system defined at middle surface of cross section;
P = pole or center of rotation;
Q = matrix of stress resultants;
q = matrix of bar strains;
r = perpendicular distance from pole to point on middle surface of section;
T_s = Saint Venant torque;
t = thickness of wall of cross section
U,V,W,= rigid body displacement of pole in the x,y,z, directions;
U_s = strain energy per unit length of bar;
w = axial displacement of point on the middle surface of the cross section
x,y,z = cartesian coordinates;
α,β,γ = dimensionless parameters locating neutral axis in cross section;
δq_i = increment of plastic bar strain;

ε_{zz} = axial strain in section;

θ = orientation of neutral axis in element of cross section;

λ = positive constant;

λ_L = dimensionless bar parameter

σ_y = yield stress of material;

σ_{zz} = axial stress in section;

τ_{zs} = shear stress in section

ϕ = rotation of cross section about pole;

ω = contour warping function

Superscript

' = differentiation with respect to z-axis

TABLE 1 - LOCATION OF NEUTRAL AXES FOR SURFACE ELEMENT 2

ELEMENT	x	y	ω
TOP FLANGE	αb	$\frac{1}{2}$ h	$-h\alpha b$
WEB	0	$-\frac{1}{2}$ h	0
BOTTOM FLANGE	$-b \leq x \leq 0$	$-\frac{1}{2}$ h	0

TABLE 2 - YIELD MECHANISMS

SURFACE ELEMENT	$\delta W'/\delta\phi''$	$\delta U''/\delta\phi''$	$-\delta V''/\delta\phi''$
1	$\frac{\alpha(1-\beta)(1-2\gamma)hb}{2[\alpha+\gamma(1-\beta-\alpha)]}$	$\frac{\alpha(1-\gamma)h}{[\alpha+\gamma(1-\beta-\alpha)]}$	$\frac{-\beta(1-\alpha)b}{[\beta-\gamma(1-\beta-\alpha)]}$
2	$-\frac{1}{2}\,h\alpha b$	0	$-\alpha b$
3	$-\frac{1}{2}\,\beta hb$	h	βb
4	0	h	0
1a	$\frac{\beta(1-\alpha)(2\gamma-1)hb}{2[\beta-\gamma(1-\beta-\alpha)]}$	$\frac{-\gamma(1-\alpha)h}{[\beta-\gamma(1+\beta-\alpha)]}$	$\frac{-\alpha(1-\beta)b}{[\alpha+\gamma(1-\beta-\alpha)]}$
2a	$-\frac{1}{2}\,h\ (1-\alpha)b$	0	$-(1-\alpha)b$
3a	$-\frac{1}{2}\,h\ (1-\beta)b$	h	$(1-\beta)b$
4a	0	h	0

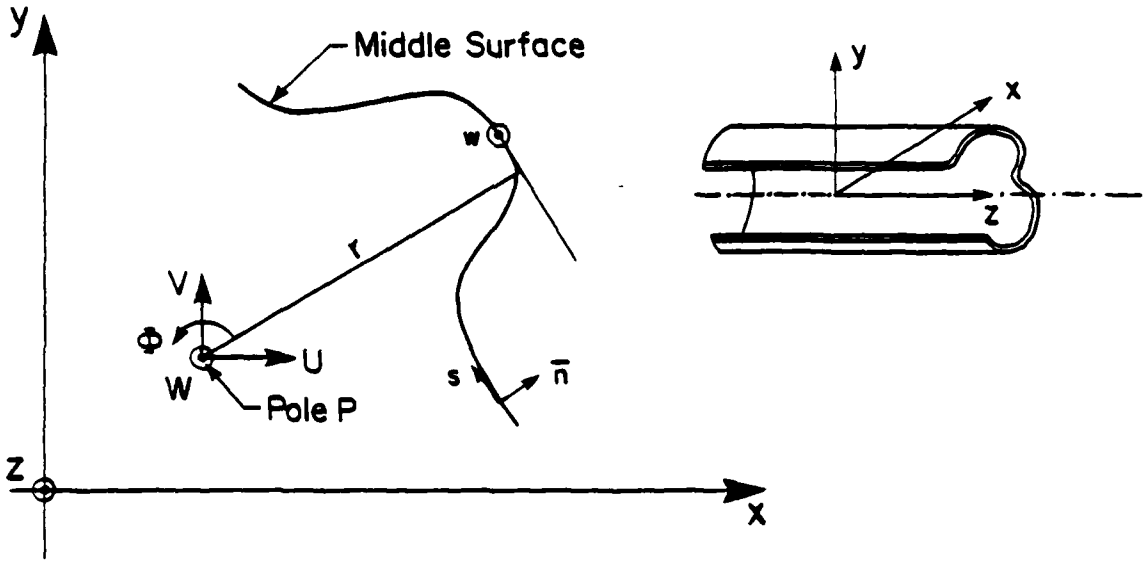


FIG.1 CROSS SECTION DISPLACEMENTS

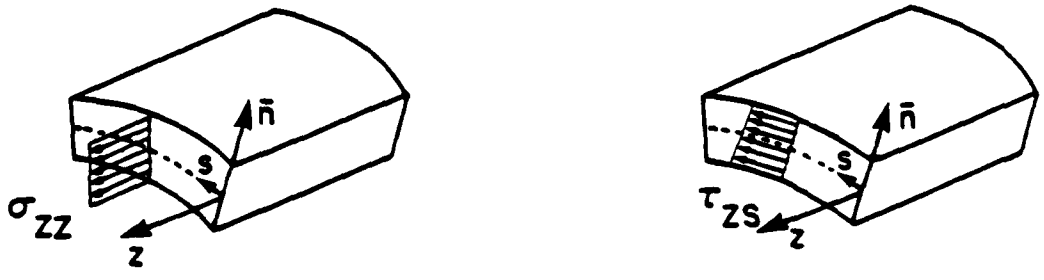


FIG.2 STRESS COMPONENTS

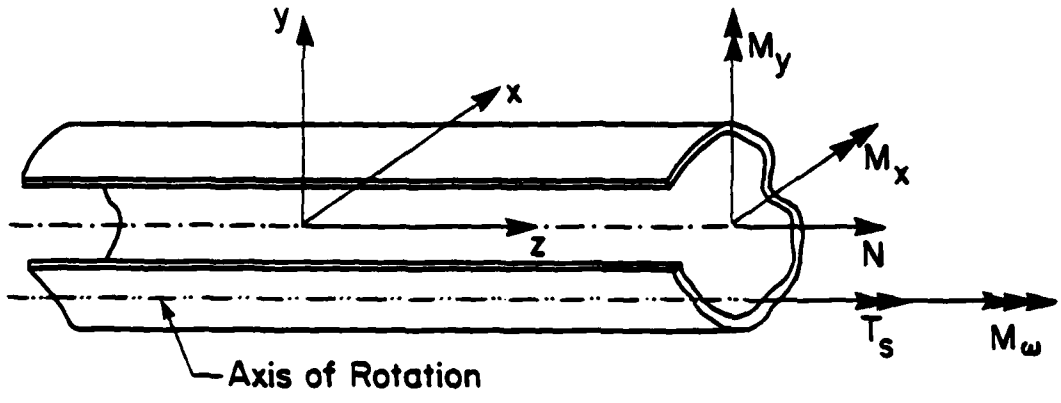


FIG. 3 STRESS RESULTANTS

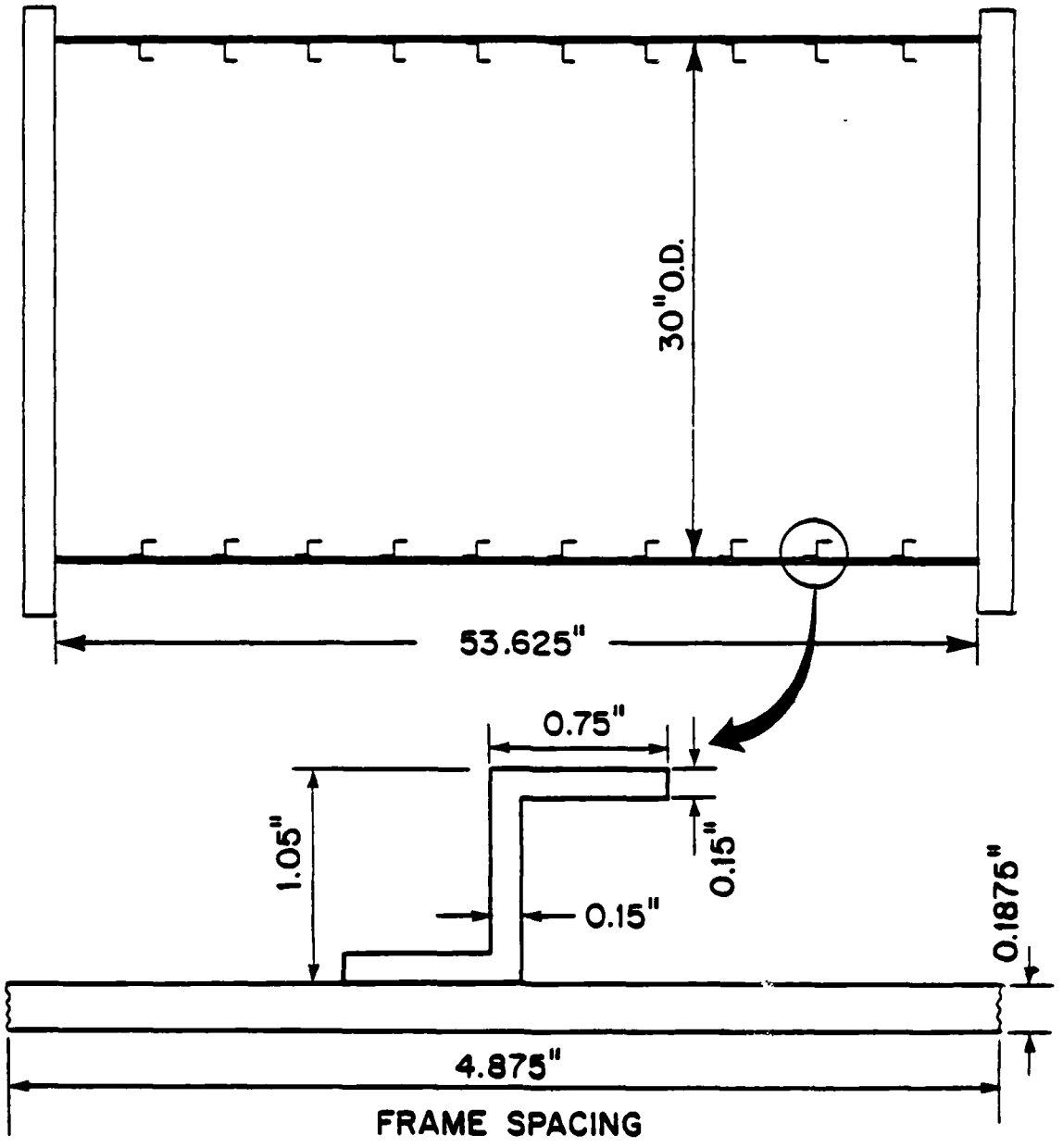


FIG. 4 Z STIFFENED CYLINDRICAL SHELL (1 in = 25.4 mm)

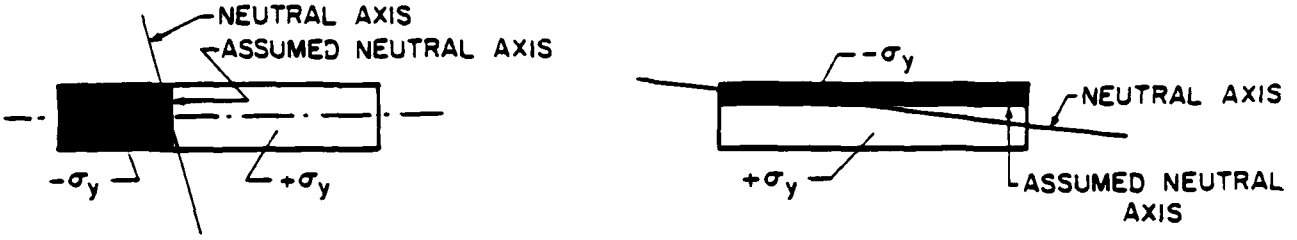


FIG. 5 ASSUMED NORMAL STRESS DISTRIBUTIONS

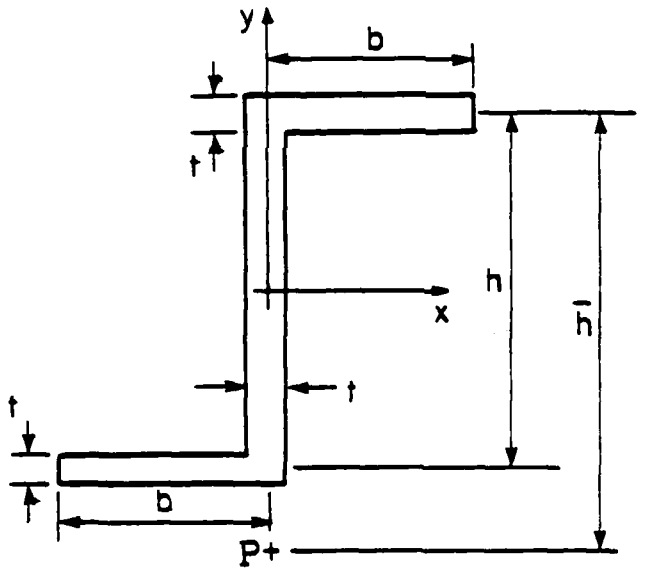


FIG. 6 Z SECTION

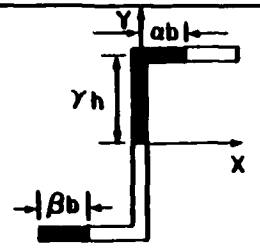
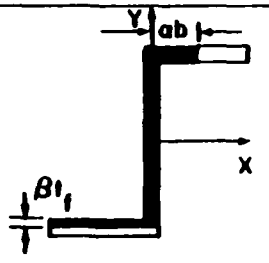
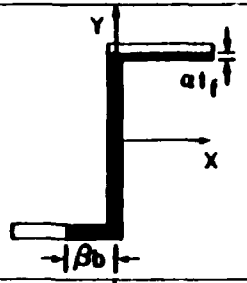
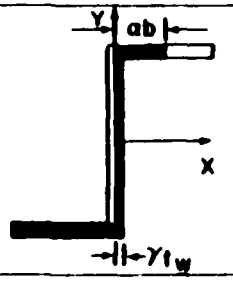
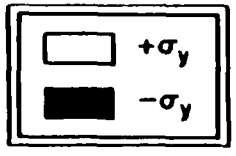
SURFACE ELEMENT	NEUTRAL AXES LOCATION	NORMALIZED STRESS RESULTANTS	YIELD SURFACE EQUATIONS
1		$n=1-\frac{2(\beta+\gamma c+\alpha)}{2+c}$ $m_x=\frac{4}{4+c}\left[\gamma c(\gamma-1)-\alpha+\beta\right]$ $m_y=\beta^2-2\beta+\alpha^2$ $m_\omega=1-2\alpha^2$	$\left[\left(\frac{2+c}{2}\right)(1-n)-\left(1-\sqrt{\frac{1}{2}(1+m_\omega)+m_y}+\sqrt{\frac{1}{2}(1-m_\omega)}\right)\right]\cdot$ $\left[\left(\frac{2+c}{2c}\right)(1-n)-\frac{1}{c}\left(1-\sqrt{\frac{1}{2}(1+m_\omega)+m_y}+\sqrt{\frac{1}{2}(1-m_\omega)}\right)-1\right]+1$ $-\sqrt{\frac{1}{2}(1+m_\omega)+m_y}-\sqrt{\frac{1}{2}(1-m_\omega)}-\left(\frac{4+c}{4}\right)m_x=0$
2		$n=-\frac{c+2(-1+\beta+\alpha)}{2+c}$ $m_x=\frac{4(\beta-\alpha)}{(4+c)}$ m_y= indeterminate $m_\omega=1-2\alpha^2$	$(2+c)n+\frac{4+c}{2}m_x+\sqrt{8(1-m_\omega)}+c-2=0$
3		$n=-\frac{c+2(-1+\beta+\alpha)}{2+c}$ $m_x=\frac{4(\beta-\alpha)}{(4+c)}$ m_y= indeterminate m_ω= indeterminate $\beta=\sqrt{\frac{1}{2}(1-m_\omega)-m_y}$	$-(2+c)n+\frac{4+c}{2}m_x-4\sqrt{\frac{1}{2}(1-m_\omega)-m_y}-c+2=0$
4		n= indeterminate m_x= indeterminate $m_y=\alpha^2-1$ $m_\omega=1-2\alpha^2$	$m_\omega+2m_y+1=0$ 

FIG. 7 YIELD SURFACE EQUATIONS

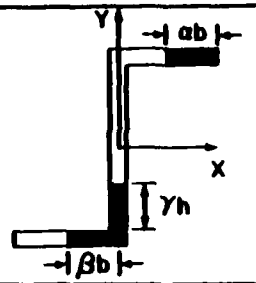
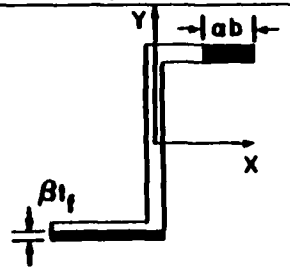
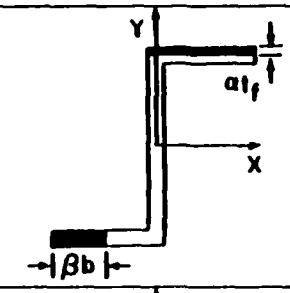
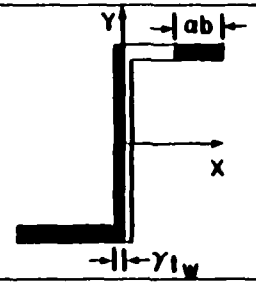
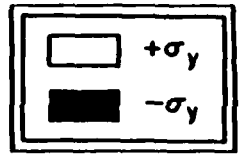
SURFACE ELEMENT	NEUTRAL AXES LOCATION	NORMALIZED STRESS RESULTANTS	YIELD SURFACE EQUATIONS
1a		$n=1-\frac{2(\beta+\gamma c+\alpha)}{2+c}$ $m_x=\frac{4}{4+c}\left[\gamma c(1-\gamma)-\alpha+\beta\right]$ $m_y=-\alpha^2+2\alpha-\beta^2$ $m_\omega=1-2\alpha(2-\alpha)$	$\left[\left(\frac{2+c}{2}\right)(1-n)-\left(\sqrt{\frac{1}{2}(1-m_\omega)-m_y}+1\div\sqrt{\frac{1}{2}(1+m_\omega)}\right)\right]\cdot$ $\left[\left(\frac{2+c}{2c}\right)(1-n)-\frac{1}{c}\left(\sqrt{\frac{1}{2}(1-m_\omega)-m_y}+1\div\sqrt{\frac{1}{2}(1+m_\omega)}\right)-1\right]+$ $\sqrt{\frac{1}{2}(1-m_\omega)-m_y}-1+\sqrt{\frac{1}{2}(1+m_\omega)}-\left(\frac{4+c}{4}\right)m_x=0$
2a		$n=1-\frac{2(\beta+\alpha)}{2+c}$ $m_x=\frac{4}{4+c}(\beta-\alpha)$ $m_y=$ indeterminate $m_\omega=1-2\alpha(2-\alpha)$	$(2+c)n+\frac{4+c}{2}m_x-\sqrt{8(1+m_\omega)}-c+2=0$
3a		$n=1-\frac{2(\beta+\alpha)}{2+c}$ $m_x=\frac{4}{4+c}(\beta-\alpha)$ $m_y=$ indeterminate $m_\omega=$ indeterminate $\beta=1-\sqrt{\frac{1}{2}(1+m_\omega)+m_y}$	$-(2+c)n+\frac{4+c}{2}m_x+4\sqrt{\frac{1}{2}(1+m_\omega)+m_y}+c-2=0$
4a		$n=$ indeterminate $m_x=$ indeterminate $m_y=-\alpha^2+2\alpha-1$ $m_\omega=1-2\alpha(2-\alpha)$	$m_\omega+2m_y-1=0$ 

FIG. 8 YIELD SURFACE EQUATIONS

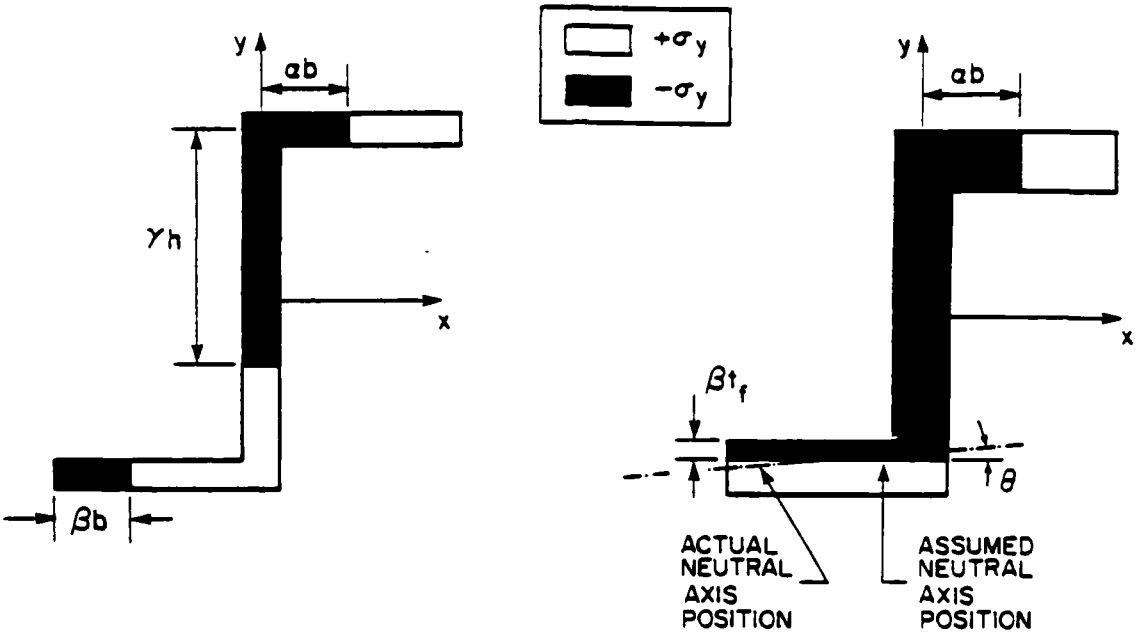


FIG. 9 ASSUMED STRESS DISTRIBUTION FOR SURFACE ELEMENT 1

FIG. 10 ASSUMED STRESS DISTRIBUTION FOR SURFACE ELEMENT 2

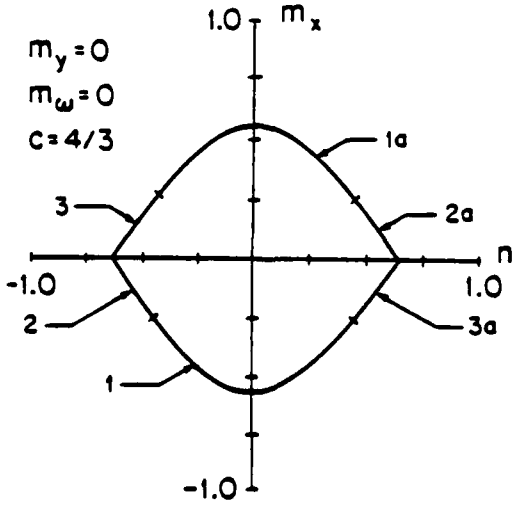


FIG. 11 Z SECTION YIELD SURFACE

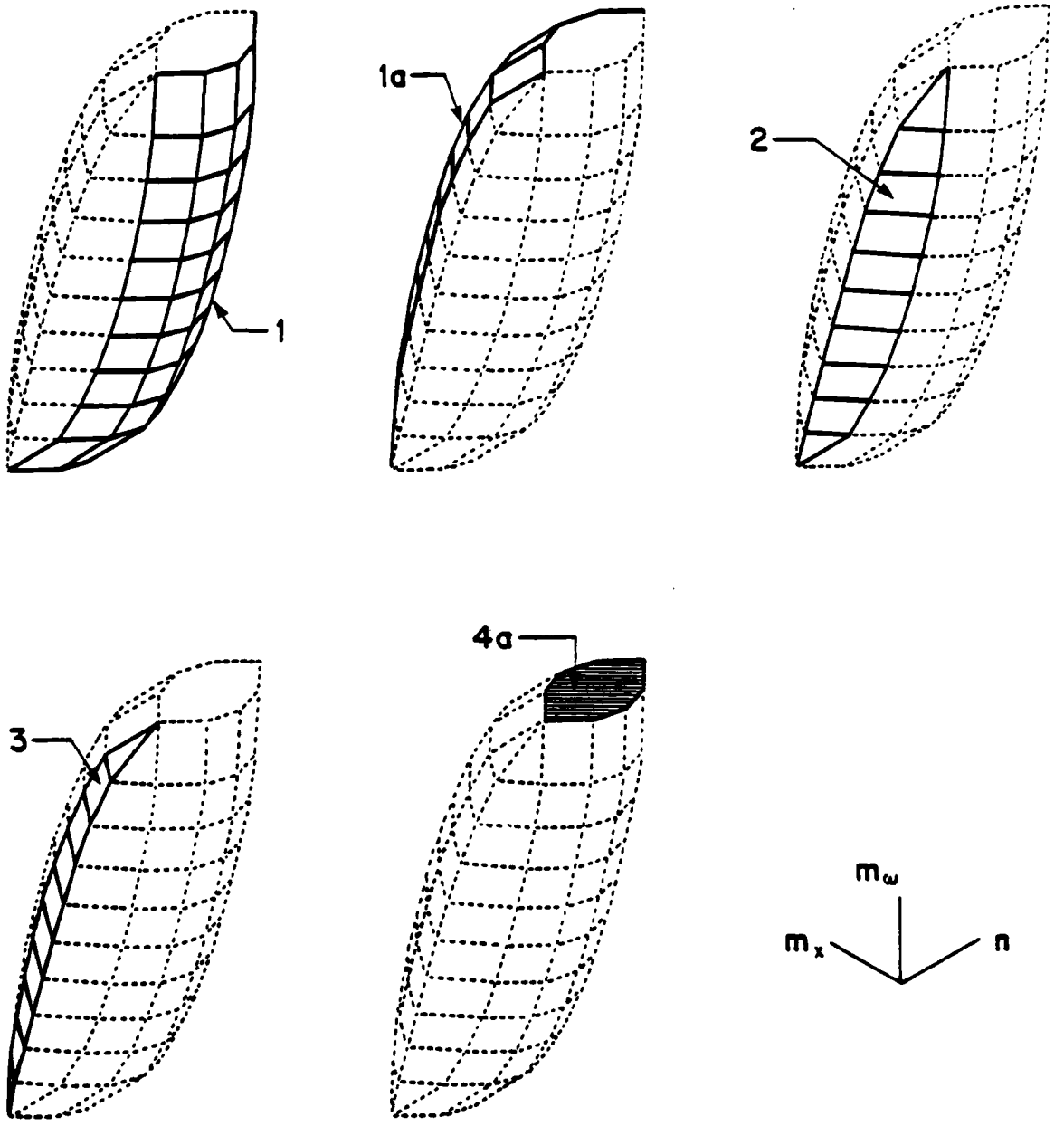


FIG. 12 Z SECTION YIELD SURFACE - m_y=0

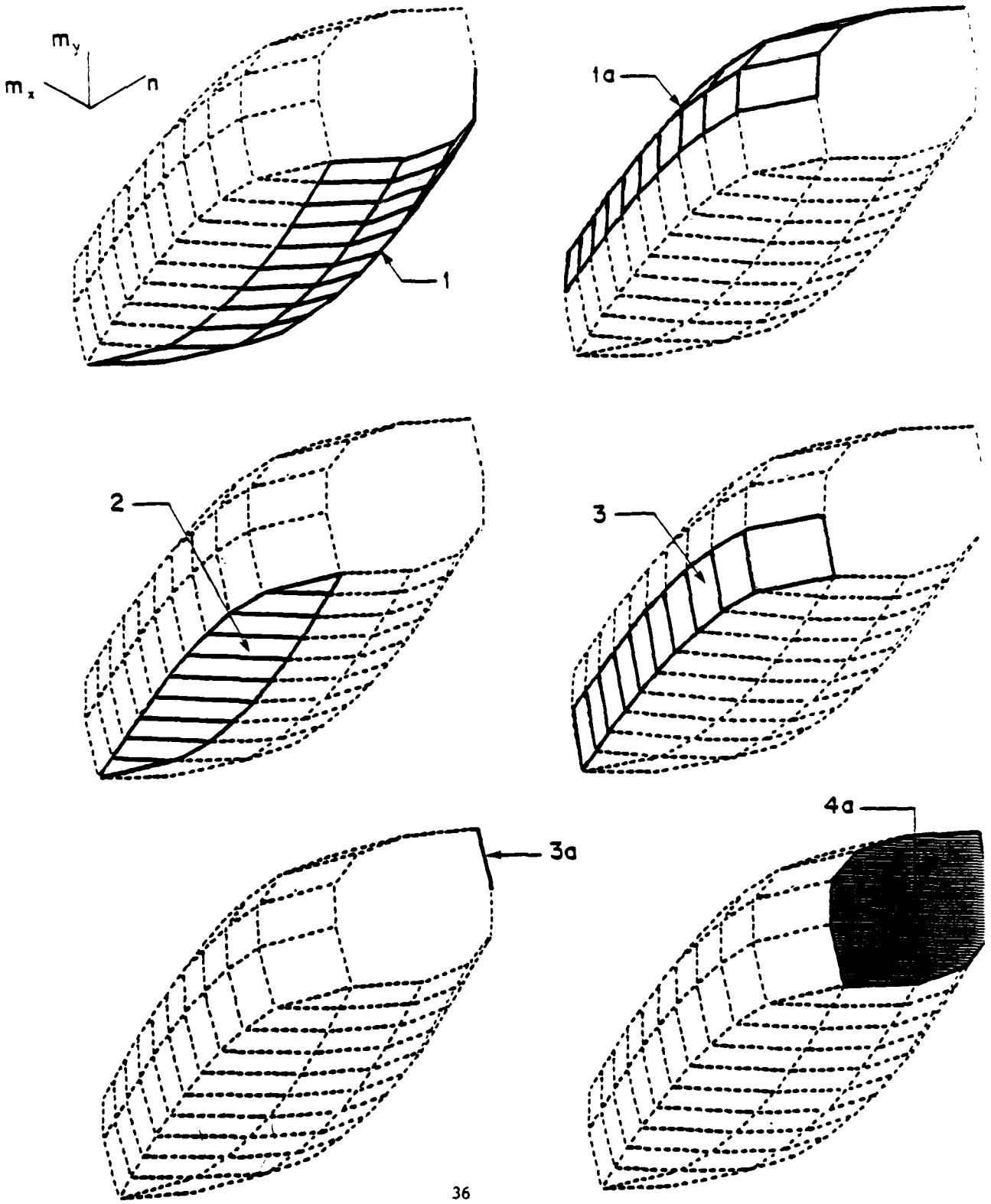


FIG. 13 Z SECTION YIELD SURFACE - $m_\omega = 0$

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